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(71) Applicant: BENTLY NEVADA CORPORATION [US/US]; 1631 Bently Parkway South, Minden, NV 89423 (US).

(72) Inventors: **BENTLY, Donald, E.**; 1631 Bently Parkway South, Minden, NV 89423 (US). **ELDRIDGE, Thomas, M.**; 1631 Bently Parkway South, Minden, NV 89423 (US). **JENSEN, Jeremy, L.**; 1631 Bently Parkway South, Minden, NV 89423 (US). **JENSEN, Raymond, V.**; 1631 Bently Parkway South, Minden, NV 89423 (US). **FOSTER, Craig, J.**; 1631 Bently Parkway South, Minden, NV 89423 (US).

(74) Agent: **DEBOO, Dennis, A.**; DeBoo & Co., Suite 900, 400 Capitol Mall, Sacramento, CA 95814 (US).

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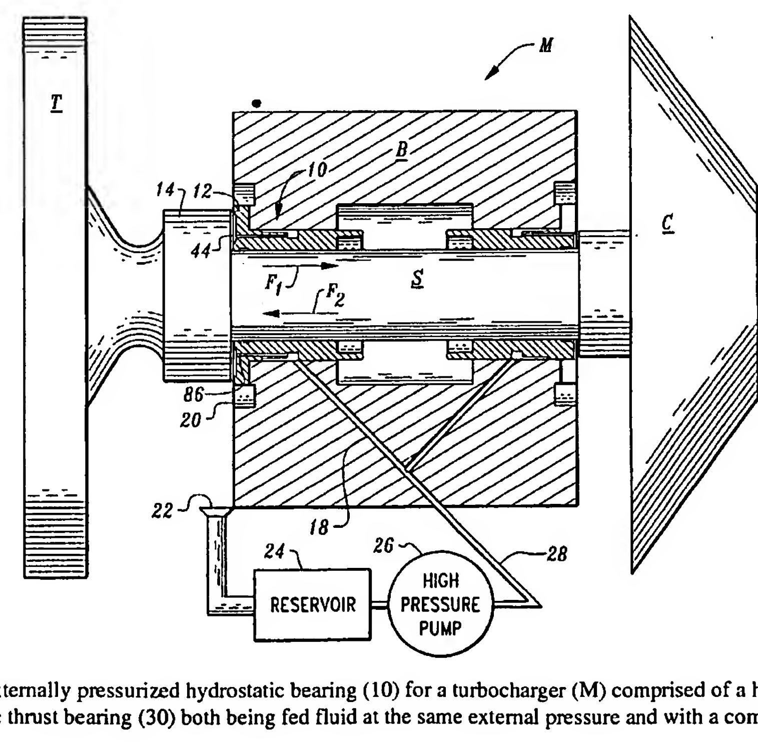
(54) Title: A HYDROSTATIC BEARING FOR USE IN A TURBOCHARGER



(57) Abstract: An externally pressurized hydrostatic bearing (10) for a turbocharger (M) comprised of a hydrostatic radial bearing (90) and a hydrostatic thrust bearing (30) both being fed fluid at the same external pressure and with a common fluid supply source (26).

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A Hydrostatic Bearing For Use In A Turbocharger

Technical Field

The instant invention relates generally to hydrostatic bearings for use in turbochargers and, in particular, to an externally pressurized hydrostatic thrust bearing, to an externally pressurized hydrostatic combination radial/thrust bearing, and to a bearing having an integral transducer mounting means all for use in a turbocharger.

Background Art

As is known in the art, turbochargers include a shaft rotatably supported within a turbocharger housing by lubricated bearings. In one form, a compressor wheel is mounted on one end of the shaft, and a turbine wheel is mounted on the other end of the shaft. Exhaust Gas traveling through the turbine wheel drives the shaft at a high speed thereby rotating the compressor wheel to compress air for delivery. As a result of operation, both radial and axial forces are produced on the shaft and one or more non-rotating radial bearings and one or more non-rotating thrust bearings are employed to resist these radial and axial forces.

Typically, turbochargers employ hydrodynamic or internally pressurized radial bearings in which rotation of the shaft "drags" the fluid in the direction of rotation to produce a fluid film or pressure wedge that displaces the shaft away from the bearing surface. The supply pressure of the fluid is not designed to contribute to the rotor dynamic properties of the bearing but rather, to merely cause adequate flow through the bearing to provide lubricant cooling. Hence, the supply pressure is typically relatively low (e.g., 15 to 20 psi).

While these hydrodynamic or internally pressurized radial bearings are currently widely employed, they work on the principle of starving the bearing by intentionally keeping it from developing a fully developed fluid film and in contrast, they rely on partially lubricating the radial bearing for increasing the stiffness and thus, stability. This is a poor and dangerous way to influence stability.

Turbochargers also typically employ hydrodynamic or internally pressurized thrust bearings in which fluid "dragged" or squeezed through, for example, a tapered bearing surface and a thrust surface of the turbine wheel causes a pressure increase and hence, a force which resists the axial thrust for obtaining a balance between the thrust load and the pressure of the bearing.

Large mechanical drags are associated with this type of thrust bearing as a result of the large areas in which fluid is being sheared between. As a result, a large amount of horsepower is lost due

to this inordinate amount of mechanical drag. Additionally, present day thrust bearings are relatively large in size, which also contributes to mechanical loss as known in the art.

Accordingly, there is a need for a radial bearing which eliminates the partially lubricated or "starvation" paradigm for providing radial bearing stiffness. Additionally, there is a need for a thrust bearing which eliminates the hydrodynamic or internally pressurized thrust bearings in which fluid is "dragged" or squeezed through opposing surfaces and which reduces the mechanical drag associated therewith. There is also a need to reduce bearing size and thus its associated mechanical loss.

Disclosure of Invention

The instant invention is distinguished over the known prior art in a multiplicity of ways. For one thing, the instant invention provides an externally pressurized hydrostatic thrust bearing, an externally pressurized hydrostatic combination radial/thrust bearing, and an externally pressurized hydrostatic bearing having an integral transducer mounting means all for use in a turbocharger.

In one form, the instant invention is comprised of an externally pressurized hydrostatic radial/thrust bearing combination for a turbocharger that includes a cylindrical construct and a disk shaped construct contiguously formed with one another and having a center bore for a shaft to pass therethrough. The cylindrical construct includes an external annular groove disposed therein and having at least one orifice providing open communication between the groove and an interior of the cylindrical construct. The disk shaped construct includes a front face having a radially extending annular pocket formed by a annular bottom surface flanked by a pair of concentrically disposed, axially extending, annular lands for defining a fluid holding means. The disk shaped construct further includes at least one capillary extending therethrough and in open communication between the annular pocket and the external annular groove for providing fluidic communication therebetween such that a single common externally pressurized fluid supply means can deliver pressurized fluid to the external annular groove wherein the pressurized fluid can be metered into an interior of the cylindrical construct for providing a fully lubricated hydrostatic radial bearing that provides increased stability and wherein the pressurized fluid can be communicated, via at least the one capillary, to the annular pocket for providing a hydrostatic thrust bearing that abates axial thrust forces in a manner that reduces mechanical drag.

The hydrostatic combination radial/thrust bearing further includes a transducer-mounting ring contiguously formed with an opposite end of the hydrostatic radial bearing, distal from the disk shaped construct. The transducer-mounting ring includes a hollow interior concentric with the center bore having an internal diameter that is greater than an interior diameter of the center bore. The transducer-mounting ring further includes at least one opened ended bore radially extending

therethrough for allowing a transducer to be operatively coupled therein for monitoring the shaft passing through the center bore and hollow interior.

Industrial Applicability

The industrial applicability of this invention shall be demonstrated through discussion of the following objects of the invention.

Accordingly, a primary object of the instant invention is to provide a new, novel and useful externally pressurized hydrostatic bearing for a turbocharger.

A further object of the instant invention is to provide an externally pressurized hydrostatic bearing for a turbocharger that reduces mechanical loss in a thrust bearing and increases stability margin in a radial bearing.

Another further object of the instant invention is to provide a hydrostatic radial bearing and a hydrostatic thrust bearing for a turbocharger that are both fed fluid from a single common externally pressurized fluid supply source wherein both bearings are simultaneously being fed with fluid that is at same pressure.

Another further object of the instant invention is to provide an externally pressurized hydrostatic thrust bearing for a turbocharger that reduces the surface area in which fluid is squeezed or sheared through thereby reducing mechanical drag and increasing horsepower.

These and other objects and advantages will be made manifest when considering the following detailed specification when taken in conjunction with the appended drawing figures.

BRIEF DESCRIPTION OF DRAWINGS

Figure 1 is a diagrammatic view of a turbocharger showing a hydrostatic bearing in section and in accordance with the instant invention.

Figure 2 is front isometric view of the hydrostatic bearing in accordance with the instant invention.

Figure 3 is a sectional view of taken along line 3 – 3 of figure 2.

Figure 4 is rear isometric view of the hydrostatic bearing in accordance with the instant invention.

Figure 5 is a sectional view of taken along line 5 – 5 of figure 4.

Figure 6 is side isometric view of the hydrostatic bearing showing removable orifices in an exploded fashion and in accordance with the instant invention.

Figure 7 is a sectional view of taken along line 7 – 7 of figure 4.

Figure 8 is rear isometric view of the hydrostatic bearing in accordance with the instant invention and shown with transducers coupled to a transducer mounting means.

Figure 9 is a sectional view of a hydrostatic thrust bearing in accordance with the instant invention and shown disposed proximate a thrust surface of a thrust member carried on a shaft.

Best Mode(s) For Carrying Out The Invention

Considering the drawings, wherein like reference numerals denote like parts throughout the various drawing figures, reference numeral 10 is directed to the hydrostatic bearing for a turbocharger according to the instant invention.

In its essence, and referring to figures 1 through 8, the hydrostatic bearing 10 is disposed in a bearing block B such that the bearing block B circumscribes the hydrostatic bearing 10. The hydrostatic bearing 10, in turn, circumscribes a shaft S of a turbocharger M. In one form, the turbocharger M includes a compressor wheel C mounted on one end of the shaft, and a turbine wheel T mounted on the other end of the shaft. Exhaust Gas traveling through the turbine wheel drives the shaft S at a high speed thereby rotating the compressor wheel C to compress air for delivery. As a result of operation, both radial and axial forces are produced on the shaft S. The shaft S further includes a thrust member 14 which is disposed thereon and located outboard of the hydrostatic bearing 10. The hydrostatic bearing 10 is comprised of a hydrostatic axial thrust bearing 30 and a hydrostatic radial bearing 90 that are both fed pressurized fluid from a single common externally pressurized fluid supply means 26 via, for example, line 28 and bearing block passageway 18. The supplied pressurized fluid is delivered to an annular groove 106 disposed in the hydrostatic radial bearing 90 wherein the pressurized fluid is metered into an interior of the hydrostatic radial bearing 90 for providing a fully lubricated hydrostatic radial bearing that provides increased stability and wherein, simultaneously, the supplied pressurized fluid is communicated, via at least one capillary 86, to an annular pocket 44 of the hydrostatic axial thrust bearing 30 for abating axial thrust forces in a manner that reduces mechanical drag. The hydrostatic bearing 10 further includes an integrally formed transducer-mounting ring 150 for mounting at least one transducer immediately adjacent hydrostatic radial bearing 90.

More specifically, and referring to figures 2 and 3, the hydrostatic bearing 10 includes a hydrostatic axial thrust bearing 30 comprised of a cylindrically shaped disk member 32. The cylindrically shaped disk member 32 includes a front face 34, a back face 36 and a body 38 extending therebetween. The body 38 includes a circumferential exterior surface 40 and a cylindrical interior surface 42 which defines a center void extending through the body of the thrust bearing along a central axis A.

The front face 34 of the thrust bearing 30 includes an annular pocket 44 having an annular bottom surface 46 radially transitioning into a pair of coaxially aligned inner and outer annularly

shaped sidewalls or lands 48, 50, respectively. The annularly shaped lands 48, 50 are coaxially aligned and radially spaced apart from one another and axially extend away from the annular bottom surface 46 of the annular pocket 44. In other words, the outer annular land 50 circumscribes the inner annular land 48 in a concentric radially spaced apart relationship as shown in figure 2.

The inner annular land 48 includes an annular exterior surface 52 defining an inner sidewall surface of the annular pocket 44 and further includes an annular interior surface 54 that is concentric with the hollow annular interior surface 42 of the body 38 of the thrust bearing 30. The annular surfaces 52, 54 may be either substantially straight or beveled. The outer annular land includes an annular interior surface 56 defining an outer sidewall surface of the annular pocket 44 and an annular exterior surface 58 that transitions into the outer circumferential surface 40 of the body 38 of the thrust bearing 30. The annular surfaces 56, 58 may be either substantially straight or beveled. The front face surfaces 49, 51 (see figure 2) of the annular lands 48, 50 provide a small surface area between the thrust member surface 12 and the thrust bearing where oil is squeezed or sheared (as shown by the directional arrows in figure 9) thereby reducing mechanical drag and thus increasing horsepower.

At least one open ended capillary 60 extends through the body 38 of the thrust bearing 30 and includes one end 62 that preferably terminates flush with the annular bottom surface 46 of the annular pocket 44 and another end 64 (please see figure 9) that is operatively coupled to a remote fluid supply means 26 for communicating pressurized fluid from the remote fluid supply means 26, through the capillary 60 and into the annular pocket 44 of the thrust bearing 30.

In one preferred form, capillary 60 is coextensive with a capillary 66 that extends through a body 92 of the radial bearing 90 thereby defining a single capillary 86 that includes one end 62 that terminates within the annular pocket 44 and another end 68 that terminates within an exterior annular groove 106 that will be delineated *infra*. Furthermore, figure 3 shows one embodiment of the thrust bearing 30 that includes; for example, a pair of these coextensive capillaries 86 that are equally spaced apart and that terminate at ends 62, 68.

The hydrostatic axial thrust bearing 30 further includes an interior, radially inwardly opening, annular groove 70 disposed in the cylindrical interior surface 42 of the axial thrust bearing 30. The interior annular groove includes a bottom circumferential surface 72 that transitions into substantially straight inner and outer side surfaces 72, 74 respectively. The side surfaces 72, 74 radially extend away from the bottom circumferential surface 72 and towards the central axis A. The interior annular groove 70 includes at least one open ended passageway 78 that radially extends through the body 38 of the thrust bearing 30 and includes an inner opened end 80 that terminates

with the bottom circumferential surface 72 and an outer opened end 82 that terminates with the outer circumferential surface 40 of the thrust bearing 30 for communicating fluid from within the cylindrical interior surface 42 to an exterior of the thrust bearing 30. In one preferred form the thrust bearing includes six equally spaced opened ended passageways 78.

Referring to figures 4 and 5, the hydrostatic bearing 10 further includes a hydrostatic radial bearing 90 comprised of a hollow cylindrical construct 92 longitudinally extending along central axis A and between a first end 94 and a second end 96. In one embodiment of the instant invention the hydrostatic radial bearing 90 is integrally formed with the hydrostatic thrust bearing 30 such that the cylindrical construct 92 extends away from the back face 36 of the hydrostatic thrust bearing 30 as shown in figure 4.

More specifically, an exterior surface 98 of the cylindrical construct 92 includes a cylindrical shoulder 100 proximate both the first end 94 of the hydrostatic radial bearing 90 and the back face 36 of the hydrostatic thrust bearing 30. The cylindrical shoulder 100 steps down at 102 and transitions into cylindrical section 104 that terminates into an exterior, radially outwardly opening, annular groove 106 disposed in the cylindrical construct 92. The exterior annular groove 106 includes an exterior annular bottom surface 108 that transitions into spaced apart, substantially straight first and second side surfaces 110, 112, respectively. The side surfaces 110, 112 radially extend away from both the bottom surface 108 and from the central axis A. The first side surface 110 transitions to the cylindrical section 104 via, for example, a first beveled edge 114 while the second side surface 112 transitions to a stepped up cylindrical section 116 via, for example, a second beveled edge 118. The stepped up cylindrical section 116 terminates at the second end 96 of the hydrostatic radial bearing 90.

As noted *supra*, and referring back to figure 3, the hydrostatic radial bearing 90 includes at least one open ended capillary 66 that axially extends through cylindrical construct 92 of the hydrostatic radial bearing 90 and includes end 68 that preferably terminates flush with the first side surface 110 of the exterior annular groove 106 which communicates fluid from remote source 26 and then through capillary 86 which extends through both the cylindrical construct 92 of the radial bearing 90 and the body 38 of the thrust bearing 30 and into the annular pocket 44. Preferably, capillary 66 disposed within the hydrostatic radial bearing 90 is integrally formed with capillary 60 disposed within the hydrostatic thrust bearing 30 such that capillary 86 is formed which includes one opened end 68 communicating with the exterior annular groove 106 disposed in the hydrostatic radial bearing 90 and another opened end 62 communicating with the annular pocket 44 disposed in the front face 34 of the hydrostatic thrust bearing 30 such that the single common fluid supply means

26 can communicate pressurized fluid to the exterior annular groove 106 which in turns feds both the hydrostatic thrust bearing 30 and the hydrostatic radial bearing 90. One or more capillaries 86 can be used such that they all connect to the supply 26 via the exterior annular groove 106.

The cylindrical construct 92 further includes a bore 120 having cylindrical interior surface 122 defining a cylindrical center void extending through the cylindrical construct 92 of the hydrostatic radial bearing 90 along the central axis A. The cylindrical interior surface 122 includes at least one pocket 124 that includes, for example, an arcuate cutaway 126 with a substantially constant radius of curvature. The actual geometrical form of the pocket 124 may be of any shape and perhaps may be evocative of a rectangle, a circle or an ellipse. In one preferred form, the cylindrical interior surface includes four circumferentially spaced pockets 124 each having an arcuate cutaway with a substantially constant radius of curvature.

Referring to figure 5, the hydrostatic radial bearing 90 further includes at least one open-ended portal 128 radially extending through the cylindrical construct 92 such that a first end 132 is substantially flush with the bottom surface 72 of the exterior annular groove 70 and a second end 132 is substantially flush with a bottom surface of one pocket 126. Preferably, each provided pocket includes and open-ended portal having the first end substantially flush with the exterior annular bottom surface of the exterior annular groove and the second end flush with each respective pocket disposed in the cylindrical interior surface of the bore.

Referring now to figures 6 and 7 and, in one preferred form of the instant invention, each portal 128 includes a removable orifice 134 that is pressed into each portal 128 for controlling fluid flow rate. Thus, different sized orifices 134 can be pressed into the portals 128 for providing different fluid flow rates and thus variable flow rate control of fluid to each individual pocket 128 of the hydrostatic radial bearing 90. In one form, the each orifice 134 includes a hollow cylindrical interior 136 that tapers 138 into a sized opening 140 in which fluid passes through before entering the cylindrical interior 136 of the hydrostatic radial bearing 90.

Referring to figure 8, and in one preferred form, the hydrostatic bearing 10 further includes an integrally formed transducer-mounting ring 150 which longitudinally extends away from the second end 96 of the radial bearing 90 along central axis A. The transducer-mounting ring 150 includes a body 152 interposed between a front end 154 and a back end 156 preferably having a beveled periphery 160. The body 152 includes a circumferential exterior surface 158 that is stepped down from second end 96 of the hydrostatic radial bearing 90. The body 152 further includes a circumferential interior surface that has a diameter that is greater than the diameter of the bore 120 of the cylindrical construct 92 of the hydrostatic radial bearing 90.

Additionally, one or more treaded bores 164 (please see figure 5) are disposed through the transducer-mounting ring 150. In one preferred form, the transducer-mounting ring 150 includes a pair of orthogonally disposed threaded bores 164 for supporting a pair of orthogonally spaced transducers 166, 168 for measuring, for example, shaft displacement.

As seen in figure 8, the increased interior diameter of the transducer-mounting ring 150 allows for one or more transducers to be threadedly coupled to each respective treaded bore 164 and extend within the hollow interior 162 of the transducer-mounting ring 150 without protruding past the diameter of the bore 120 of the cylindrical construct 92 of the hydrostatic radial bearing 90 thereby precluding contact with shaft S extending therethrough as shown in figure 1.

Referring to figure 9, the operation of the thrust bearing 30 will now be explored in greater detail. Passageway 28 from fluid supply source or high-pressure pump 28 feeds one or more capillaries 60 directly or one or more capillaries 86 via the exterior annular groove 106 (please see figures 1 and 3) with a stream of externally pressurized fluid which is restricted by these capillaries and thus drops to a lower pressure inside annular pocket 44. The pressurized fluid within annular pocket 44 escapes past the inner and outer lands and returns to the source or pump where it is again pressurized. The areas between the lands and the thrust surface also act as restrictors that cause a pressure drops.

Now in operation, the pressure of fluid within the area of the annular pocket produces a bearing force which coacts with the thrust surface 12 of the thrust member 14 so that when a certain thrust force is created by the thrust member and if that force is greater than the existing bearing force it causes the thrust member to move closer to the front face of the thrust bearing 30 thereby closing, or making smaller, the area between the lands and the thrust surface. This results in making the restriction greater and abating fluid from escaping from the pocket. As a result, the pressure of the fluid within the pocket area increases thereby increasing the bearing force on the thrust member and countering the thrust force for regaining a balance between the thrust member and the thrust bearing 30. In one analogy the lands can be thought of as leaky seals for returning fluid.

In one example wherein the thrust bearing 30 must be able to coact with a 4,000 pound thrust load the fluid supply source or high-pressure pump 28 may be set up to deliver fluid pressure at 850 psi and the thrust bearing 30 may have the following design parameters: an annular pocket pressure of approximately 700 psi; a pair of capillaries each one inch in length and having a forty thousandths of an inch diameter; a thrust bearing outside diameter of 3.5 inches; each individual land having a thickness of 0.17 inches; a interior diameter of 1.8 inches; a pocket depth of about one half to about

one inches; and an initial gap between the lands and the thrust surface of approximately two thousandths of inch; and a flow rate through the baring of about ten gallons per minute.

In summary, the instant invention provides a novel design of both a hydrostatic radial bearing 90 and a hydrostatic thrust bearing 30 that can be both fed fluid at the same external pressure and with the same common fluid supply source and then transform this externally pressurized fluid for providing a fully lubricated hydrostatic radial bearing 90 that provides increased stability and an hydrostatic axial thrust bearing 30 that abates axial thrust forces in a manner that reduces mechanical drag. The fluid supply means or pump 26 can be a variably controlled supply of fluid for providing externally pressurized fluid at variable pressure rates.

Moreover, having thus described the invention, it should be apparent that numerous structural modifications and adaptations may be resorted to without departing from the scope and fair meaning of the instant invention as set forth hereinabove and as described hereinbelow by the claims.

CLAIMS

I Claim:

Claim 1 – A hydrostatic bearing for a turbocharger, comprising in combination:

a hydrostatic radial bearing;

a hydrostatic thrust bearing;

means for feeding both said hydrostatic radial bearing and said hydrostatic thrust bearing with fluid at the same external pressure and with the same fluid supply source.

Claim 2 – The hydrostatic bearing of claim 1 wherein said hydrostatic thrust bearing is contiguously formed with said hydrostatic radial bearing.

Claim 3 – A hydrostatic bearing for a turbocharger, comprising in combination:

a cylindrical construct having a hollow interior defining a center void extending along a central axis;

said cylindrical construct including an external channel and including at least one orifice in open communication between said hollow interior and said external channel;

a disk shaped construct having a hollow interior defining a center void extending along the central axis;

said disk shaped construct including a pocket disposed within a face of said disk shaped construct at a location substantially perpendicular to the central axis;

said pocket in open communications with said external channel via at least one capillary extending therebetween;

means for supplying an externally pressurized flow of fluid to said external channel wherein the pressurized fluid is metered into said hollow interior of said cylindrical construct via said at least one orifice and wherein the pressurized fluid is also delivered to said pocket via said capillary for retarding both radial and axial forces associated with a rotating shaft extending through said hollow interiors of said cylindrical and disk shaped constructs.

Claim 4 – A hydrostatic axial thrust bearing for a turbocharger, comprising in combination:

a disk shaped member having a front face, a rear face and body extending therebetween;

a pocket formed within said front face of said disk shaped member for receiving pressurized fluid via a capillary extending through said body and in open communication with said pocket for substantially filling said pocket with fluid;

said pocket of fluid axially coacting with a thrust surface of a rotating thrust member for retarding axial movement of the rotating thrust member thereby providing a hydrostatic axial thrust bearing.

Claim 5 - The hydrostatic axial thrust bearing of claim 4 wherein said pocket includes a annularly shaped bottom surface bounded by an axially extending annularly shaped inner land circumscribe by an axially extending annularly shaped outer land disposed in a concentric radially spaced apart relationship.

Claim 6 – A hydrostatic bearing for a turbocharger, comprising in combination:

a hydrostatic radial bearing having a hollow construct longitudinally extending between two ends;

a transducer mounting means integrally formed with and longitudinally extending away from at least one of said two ends of said hydrostatic bearing for supporting at least one transducer monitoring shaft precession of a shaft passing through said hollow construct of said hydrostatic radial bearing at a location adjacent said hydrostatic radial bearing.

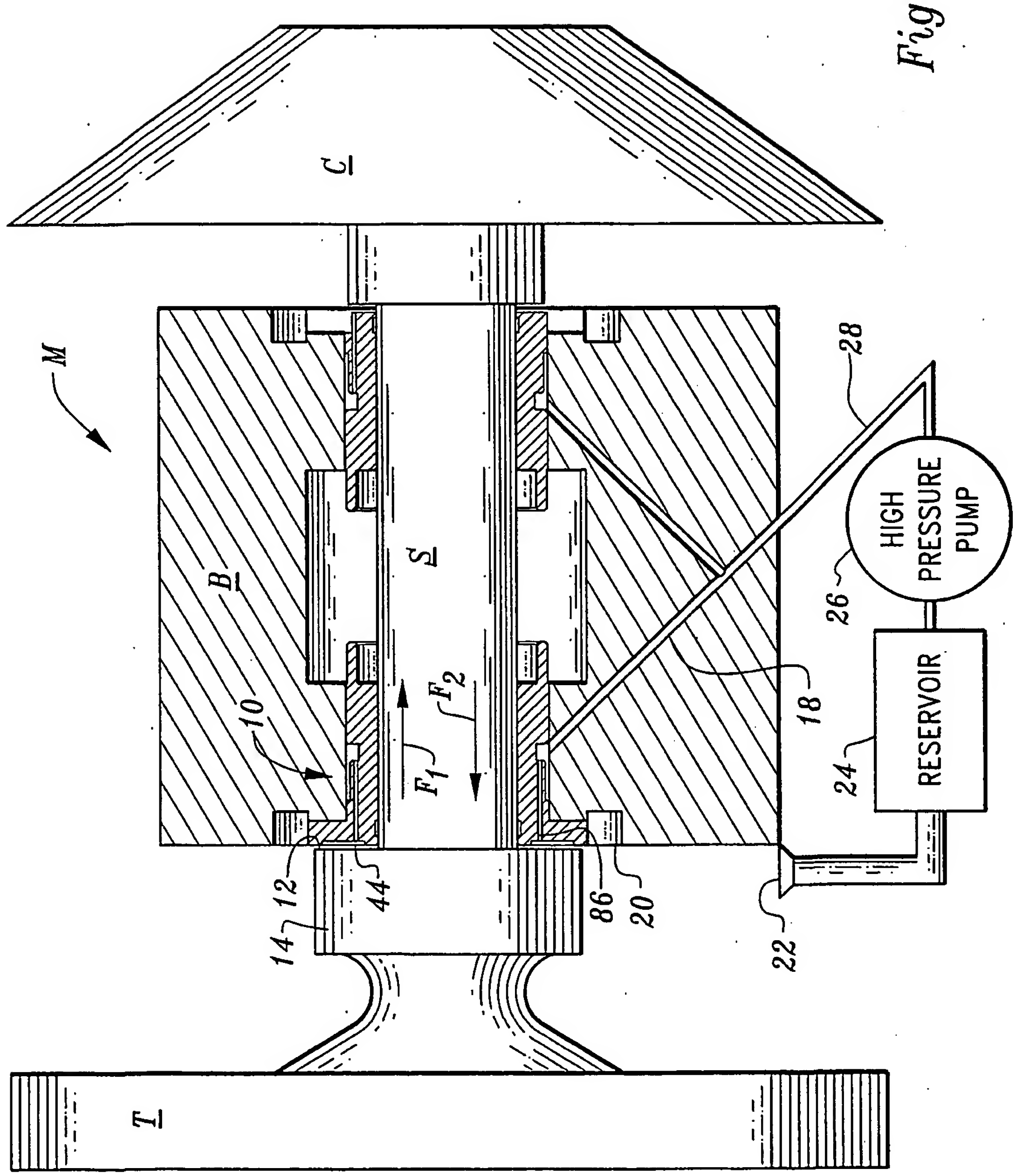
Claim 7 – The hydrostatic bearing of claim 6 wherein said transducer mounting means is ring shaped and includes a hollow interior bounded by an interior surface having a first diameter.

Claim 8 – The hydrostatic bearing of claim 7 wherein said hollow construct of said hydrostatic radial bearing is bounded by an interior surface having a second diameter that is less than said first diameter such that said at least one transducer is supported by said transducer mounting means while partially protruding past said first diameter without protruding past said second diameter of said hollow construct of said hydrostatic bearing such that said at least one transducer remains out of contact with the shaft passing through said hollow construct of said hydrostatic radial bearing.

Claim 9 – The hydrostatic bearing of claim 6 further including a hydrostatic thrust bearing integrally formed with said hydrostatic radial bearing at one of said two ends opposite said integrally formed transducer mounting means.

Claim 10 – The hydrostatic bearing of claim 9 further including means for feeding both said hydrostatic radial bearing and said hydrostatic thrust bearing with fluid at the same external pressure and with the same fluid supply source.

Fig. 1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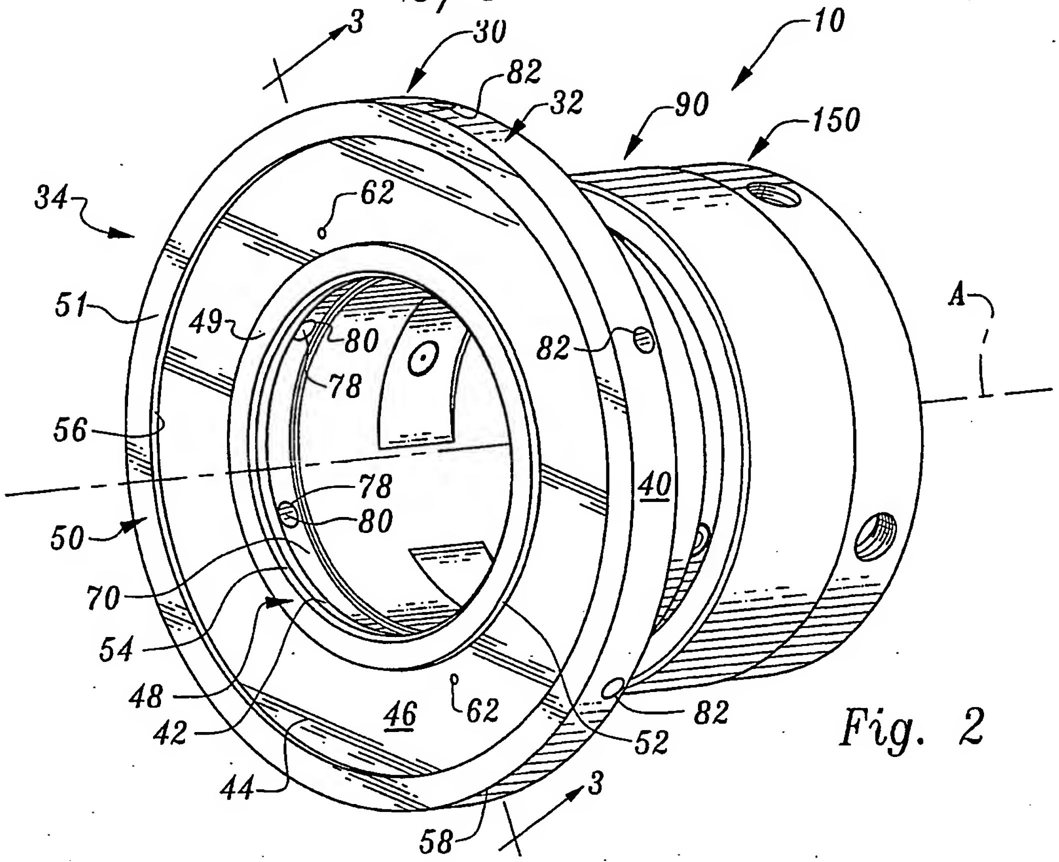
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Fig. 2

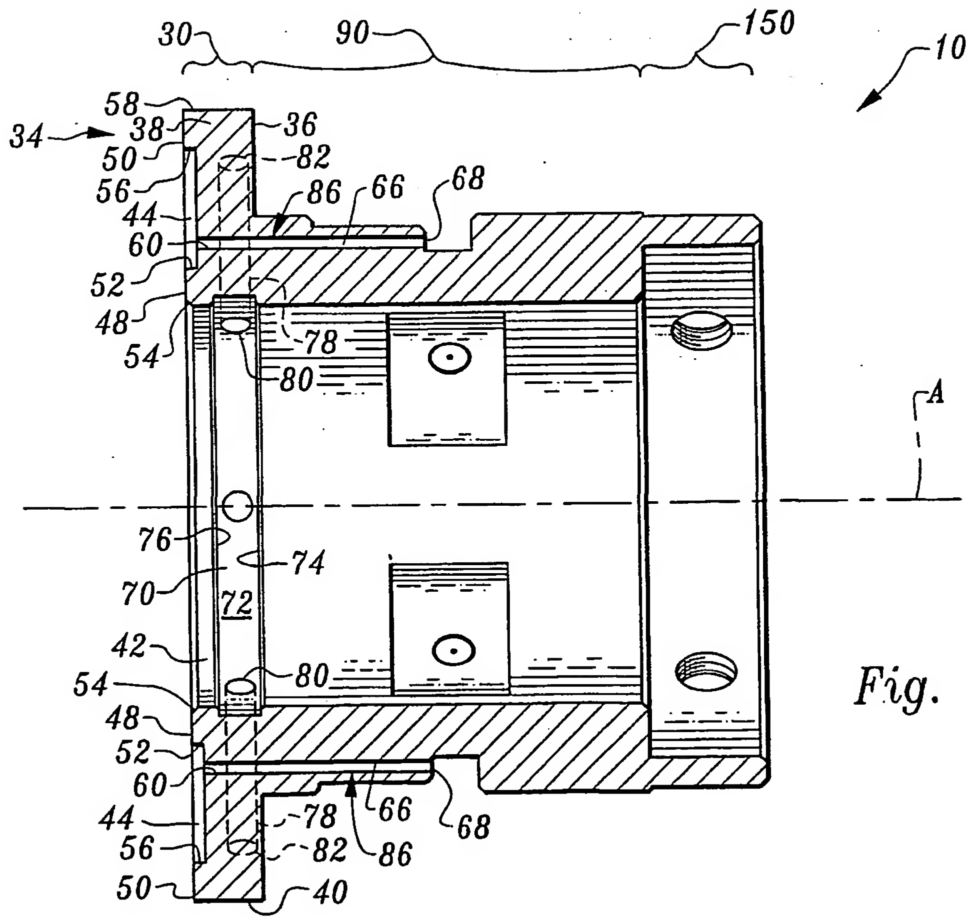
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Fig. 3

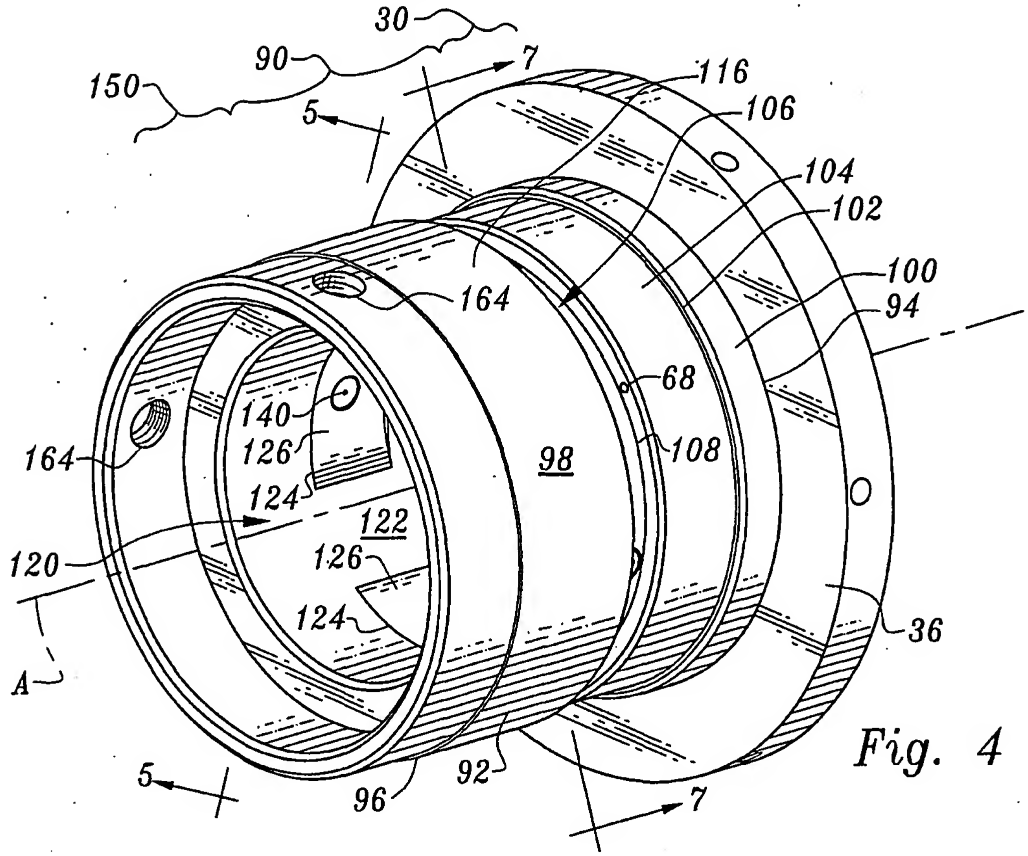
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Fig. 4

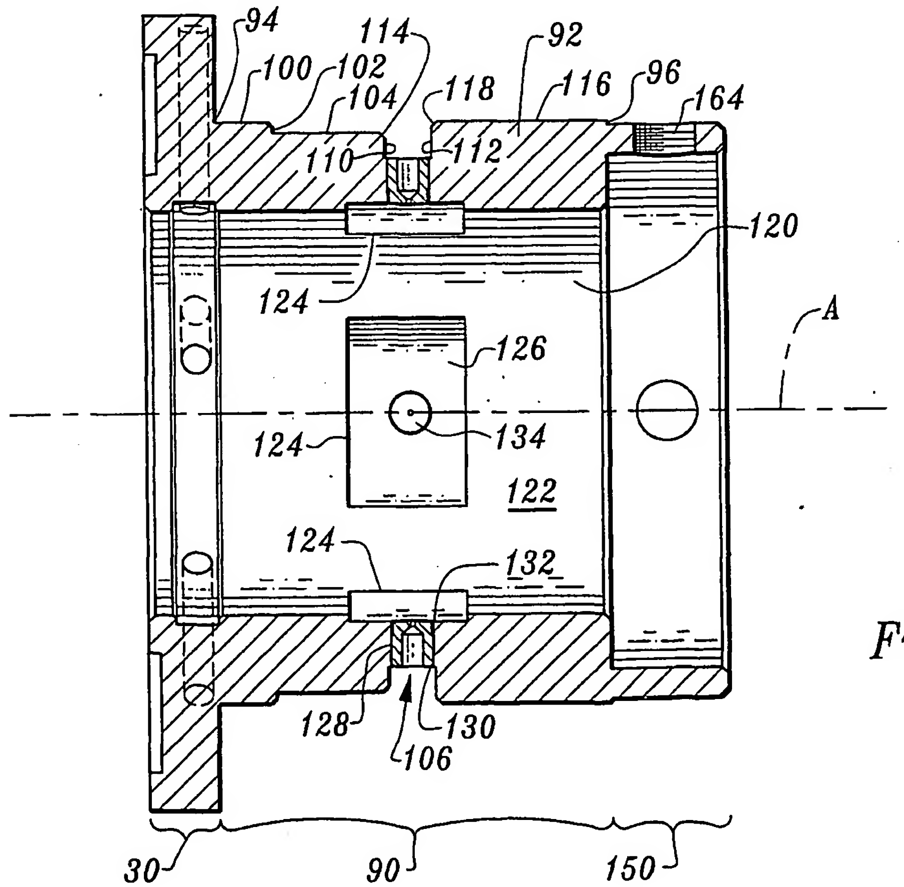
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Fig. 5

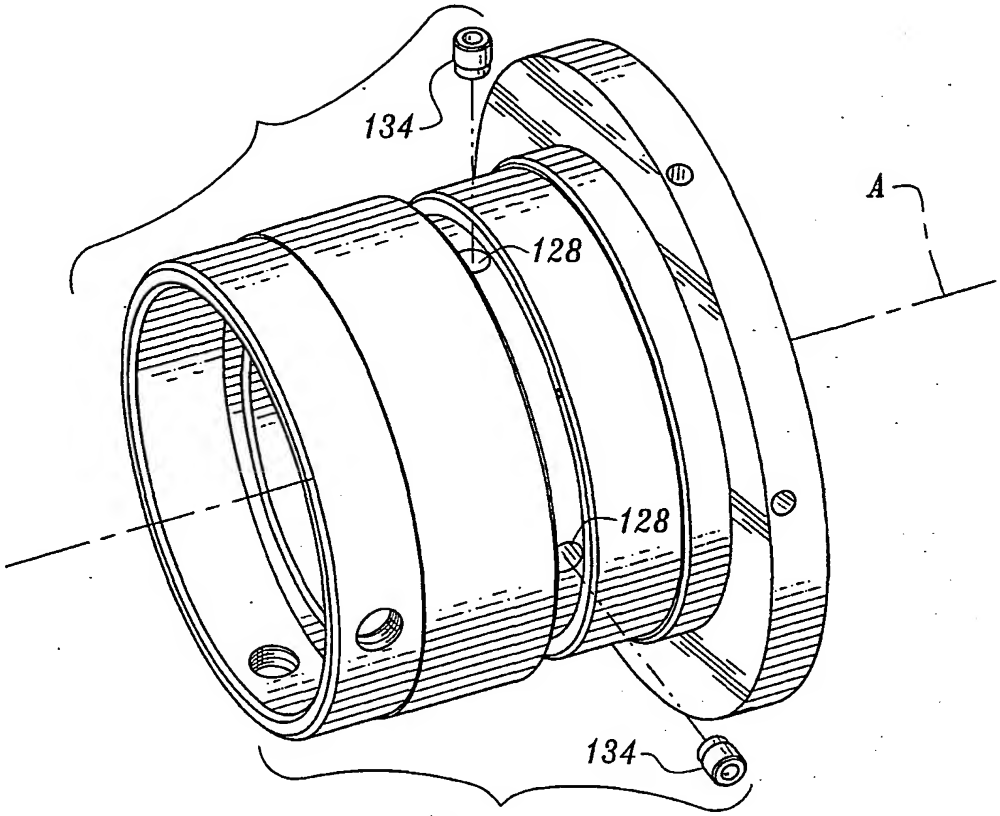
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Fig. 6

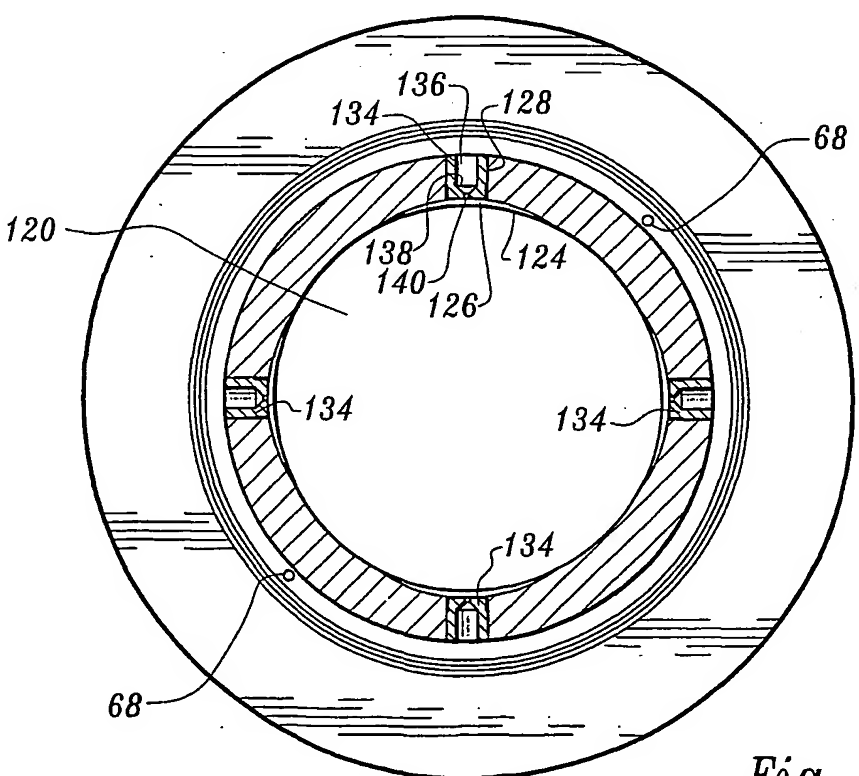
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Fig. 7

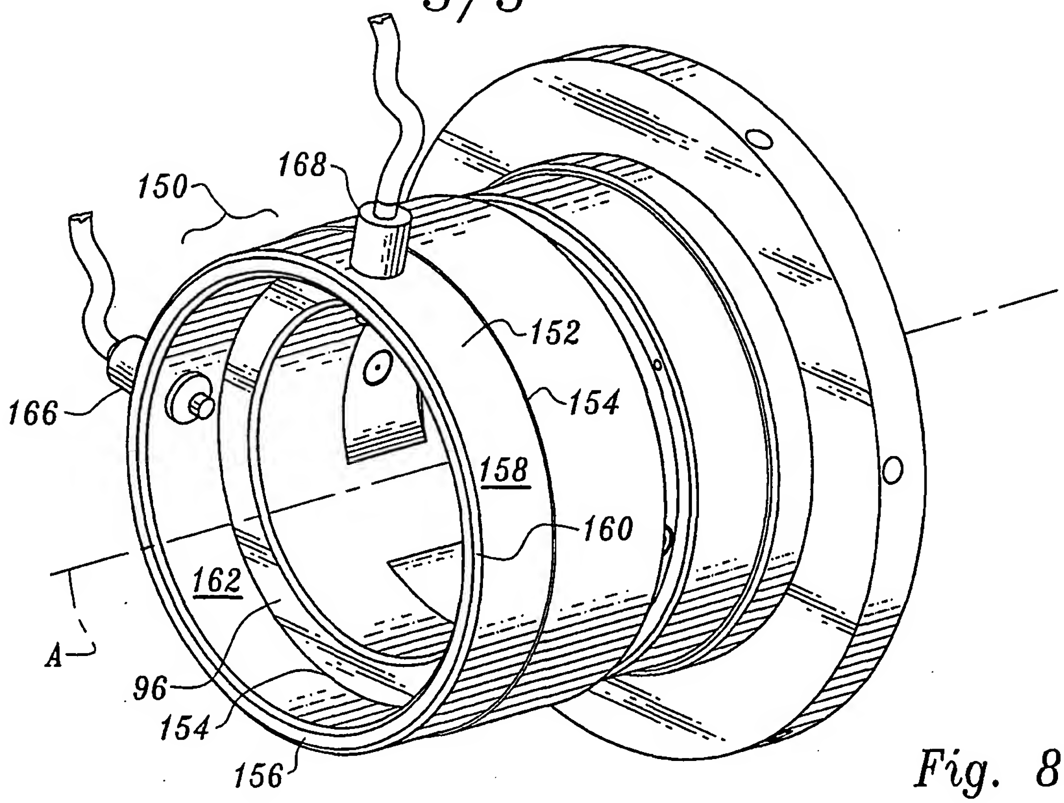

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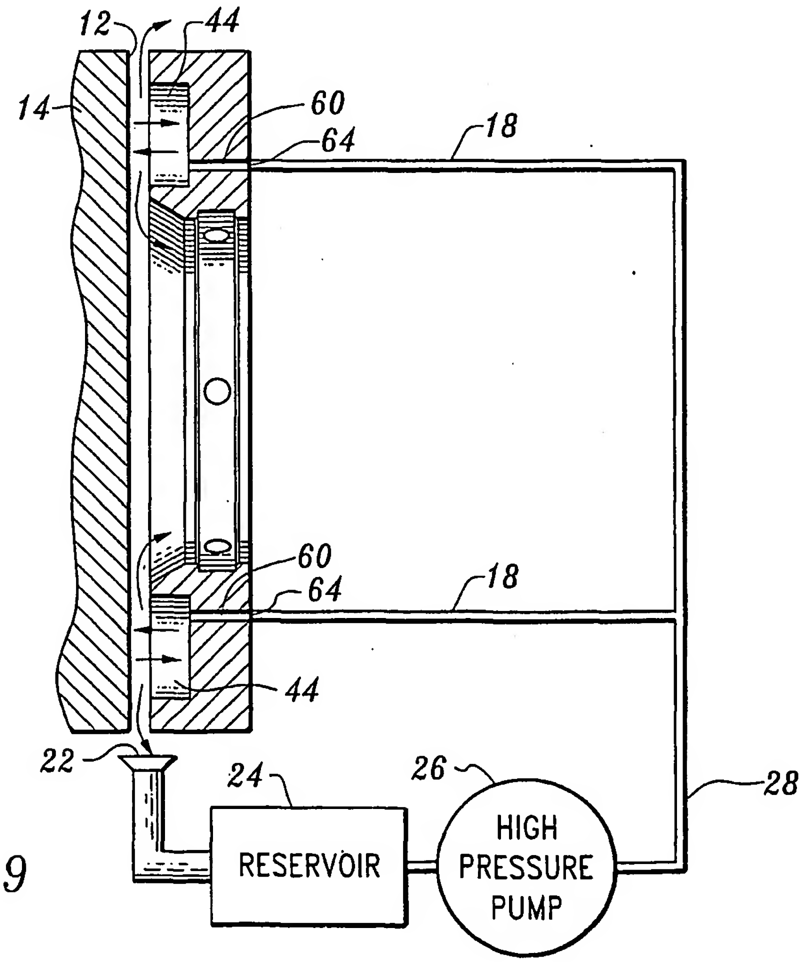
Fig. 8



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RESERVOIR
HIGH
PRESSURE
PUMP

Fig. 9

INTERNATIONAL SEARCH REPORT

International application No.
PCT/US01/21449

A. CLASSIFICATION OF SUBJECT MATTER
IPC(7) : F16C 32/06
US CL : 384/100
According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
U.S. : 384/100, 107, 111, 113, 114, 118, 120, 121

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category *	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 1,906,715 A (Penick) 02 May 1933 (05.02.1933), see all.	1-4
X	US 3,466,952 A (Greenberg et al.) 16 September 1969 (16.09.1969), see all.	1-4
X	US 3,563,618 A (Ivanov) 16 February 1971 (16.02.1971), see all.	4, 5
X	US 3,617,102 A (Wada et al.) 02 November 1971 (02.11. 1971), see all.	1, 2
X	US 4,090,743 A (Suzuki et al.) 23 May 1978 (23.05.1978), see all.	1-5
X	US 4,327,592 A (Fincke) 04 May 1982 (04.05.1982), see all.	1, 2, 4
X	US 4,573,808 A (Katayama) 04 March 1986 (04.03.1986), see all.	1, 2
X	US 4,836,692 A (Hagen) 06 June 1989 (06.06.1989), see all.	1, 2
X	US 4,915,510 A (Arvidsson) 10 April 1990 (10.04.1990), see all.	4, 5

☒ Further documents are listed in the continuation of Box C. ☐ See patent family annex.

* Special categories of cited documents:

"A" document defining the general state of the art which is not considered to be of particular relevance

"E" earlier application or patent published on or after the international filing date

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"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

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"&" document member of the same patent family

Date of the actual completion of the international search 27 September 2001 (27.09.2001)	Date of mailing of the international search report 25 OCT 2001
Name and mailing address of the ISA/US Commissioner of Patents and Trademarks Box PCT Washington, D.C. 20231 Facsimile No. (703)305-3230	Authorized officer Thomas R. Hannon Diane Smith Telephone No. (703) 308-2168

International application No.

PCT/US01/21449

INTERNATIONAL SEARCH REPORT

C. (Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

Category *	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 5,073,037 A (Fujikawa et al.) 17 December 1991 (17.12.1991), see all.	1, 2
X	US 5,219,447 A (Arvidsson) 15 June 1993 (15.06.1993), see all.	4
X	US 5,553,948 A (Ito) 10 September 1996 (10.09.1996), see all.	1, 2
X	US 5,645,354 A (Heinzl et al.) 08 July 1997 (08.07.1997), see all.	1, 2